HANDHELD CALCULATOR PROGRAMS for ROTATING EQUIPMENT DESIGN

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To my wife, Ida Fielding

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Preface

In the design of rotating equipment it is current practice to employ sophisticated high-speed computers to analyze designs in order to ensure mechanical integrity and performance capability. This analysis is usually performed at a point in the design cycle when all of the major variables such as bearing span, flowpath geometry, etc., have been fixed, and consequently it is important that these parameters be determined to a close approximation at the preliminary design stage. The initial phase of the design cycle is highly dependent upon the skill and experience of the individual design engineer, especially if the equipment to be designed is new and unique. However, whether the starting point of a design is a blank piece of paper or a previous similar design, the design engineer needs to be able to evaluate rapidly the various options available to ensure that the best compromise is made between the factors influencing the choice of parameters. The collection of programs presented in this book is intended, in part, to fulfill this need. A major problem in coding several of the programs has been the limited storage capacity of the TI 59, which in certain instances has led to compromises. An example of this is the manual search required for the natural frequencies in the critical speed and blade vibration programs; however, these compromises do not detract from the usefulness and simplicity of the programs, which are accurate, powerful analytical tools.

The programs are divided into three broad sections, namely, Vibration Analysis, Mechanical Design Analysis, and Fluid Dynamic Analysis.

Each program has been written to be self-contained with a brief theory section. The theory section will permit the program user to ensure the applicability of a program to a particular problem without the need to refer to other references.

In writing the many programs for this volume there has been some concern relative to the correctness of the logic, and wherever possible calculator solutions have been checked against analytic solutions or hand calculations. In the case of the lateral critical speed, torsional critical speed, and blade flexural vibration programs, sample case solutions have been verified against the output from large-scale computer programs. Many of the problems used to verify the programs have been given in the text and should assist in checking the programs to ensure that the correct logic has been entered into the calculator prior to recording for permanent storage. Several of the programs employ iterative routines, and while all of these routines have been checked for convergence with sample cases, it is possible that certain combinations of variables could preclude convergence, particularly if the parameters chosen lie far outside the expected range for the particular problem under consideration. In the event of nonconvergence, provision has been made, in many programs, for the iterative routines to be stepped through manually. This option should materially assist in determining the reasons for nonconvergence.

Finally, the book is intended to provide the design engineer with the capability of rapidly evaluating preliminary designs, with a minimum of hand calculations, and should be an invaluable aid in the design of rotating equipment.

Leslie Fielding

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VIBRATION ANALYSIS

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Lateral Critical Speed Program

INTRODUCTION

An important consideration in the design of rotating machinery is the placement of the system critical speeds, and good dynamic design requires that critical speeds be removed from the speed ranges at which sustained operation takes place. The rotor influences the critical speeds through the distribution of mass, inertia, and stiffness properties, all of which affect the mode shapes and natural frequencies of the system. Gyroscopic effects of shafts may usually be ignored; however, disk gyroscopic effects must be included in the analysis if disk inertias are large, since critical speeds may be raised considerably by high disk inertias. Bearings influence the mode shapes and natural frequencies by their radial properties of dynamic stiffness and dynamic damping. It is usual to express these properties by the use of eight stiffness and damping coefficients that enable the anisotropic characteristics of the bearings to be taken into account.

At the preliminary design stage the location of critical speeds and the effects of bearing and structural flexibility may be approximated by considering the rotor to be represented by a number of discrete masses joined by massless elastic shaft elements, and the bearings to be represented by a single spring coefficient instead of the eight actual coefficients. The program of this chapter has been written to calculate rotor critical speeds based upon these assumptions and represents a powerful tool for use at the preliminary design stage.

PROGRAM DESCRIPTION

The method of solution used in the program is the Prohl-Myklestad method, which essentially obtains a solution to the Bernoulli-Euler equation by trial and error. This method was chosen since it avoids the necessity of having to assume a mode shape, which is a requirement of the Rayleigh, Rayleigh-Ritz, and Stodola methods. Also, higher natural frequencies can be obtained without difficulty. In the mathematical model, a

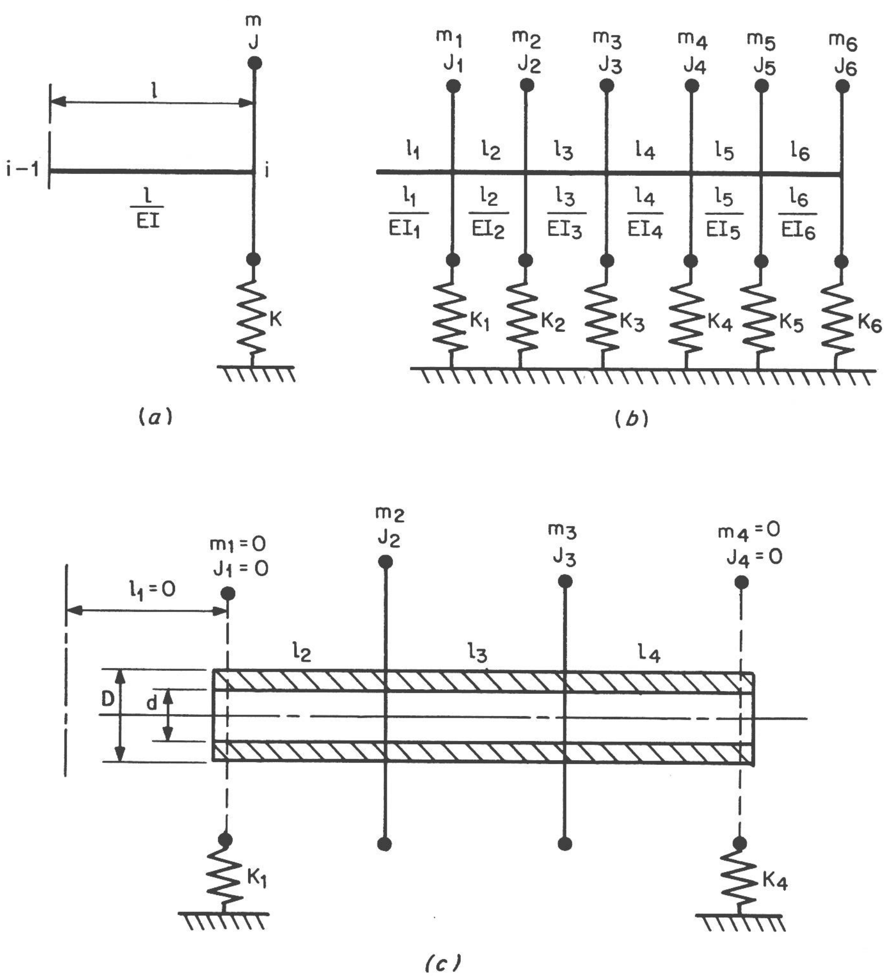

FIGURE 1.1 Use of shaft elements to describe shaft and disk systems: (*a*) Single mass element; (*b*) most complicated system program will accept; (*c*) a two-mass system

TABLE 1.1

Element number	D	d	l	W*	J	K
1	1	0	0	0	0	K_1
2	D_2	d_2	l_2	W_2	J_2	0
3	D_3	d_3	l_3	W_3	J_3	0
4	D_4	d_4	l_4	0	0	K_4

* The input required is weight, from which the program will calculate mass.

rotor section is represented by a transfer matrix which is a combination of an elastic field with a single-point mass and a linear spring support. Disk rotary inertia is included in the matrix. Figure 1.1*a* is a diagram showing the representation of a single shaft element. The maximum number of shaft elements that the calculator will accept is six, and Fig. 1.1*b* is a diagram showing the most complicated configuration that can be devised, utilizing all six elements. Figure 1.1*c* shows how the elements are used to describe a two-mass system on flexible bearings.

In reference to Fig. 1.1*c*, four elements are needed to describe the rotor system. The numerical values of the various parameters may be tabulated as shown in Table 1.1.

In the model, element 1 is used to define the left-hand bearing spring rate. All other parameters are 0; however, since the term l/EI is calculated in the program, it is necessary that EI have a value other than 0 to avoid dividing by 0 and thus causing the calculator to overflow. This can be accomplished by setting $D = 1$. Since l is equal to 0, the term l/EI will equal 0 even though D is equal to 1. Elements 2 and 3 define shaft lengths l_2, l_3; disk masses m_2, m_3; and disk inertias J_2, J_3. Spring rates K_2, K_3 are 0. Element 4 defines shaft length l_4 and bearing spring rate K_4, with m_4 and J_4 being 0. Since data is entered into the calculator in the sequence given in Table 1.1, a similar table should be drawn up for each case to be analyzed since this will minimize input errors. The above model assumes that the mass of each disk is concentrated at its center of gravity and that the rotor stiffness is represented by the stiffness of the shaft sections between disks. The mass of each shaft section has been assumed equal to 0; however, if this mass is significant relative to the disk masses, half of the shaft weight should be added to the disk at each end of the shaft section. In the model this would result in m_1 and m_4 having values equal to the half-shaft masses of l_2 and l_4, respectively, and masses m_2 and m_3 would increase to account for the half-shaft masses of l_2, l_3, and l_4.

The program is recorded on three magnetic cards. The first card enters the data, the second card allows the trial-and-error search for critical speeds, and the third card calculates the critical speed mode shape once the critical speed has been determined. The input-output sequences of the

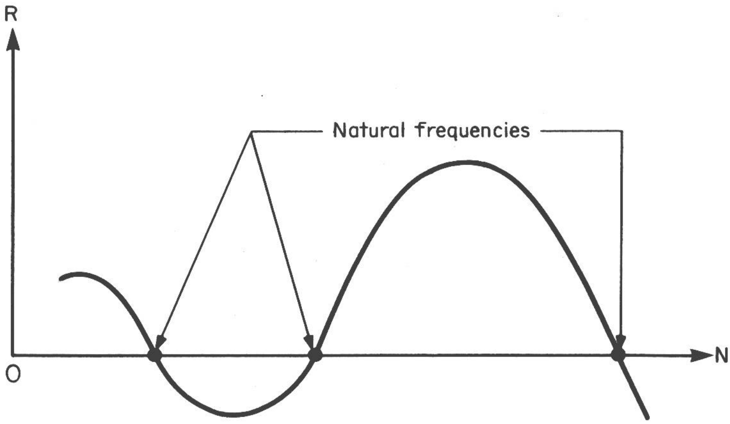

FIGURE 1.2 Natural frequency determination plot of matrix residual as a function of rotor speed

programs of cards 1 and 3 are explained fully in the input-output table and require no further explanation. The program of card 2, which is the critical speed search program, requires further comment.

The data input to the first program includes an initial guess at a critical speed and a value by which this guess is to be incremented. The program of the second card is initiated by pressing 2nd E′, whereby the program will calculate a residual R of a 2×2 matrix defining the boundary conditions. The existence of a natural frequency requires that this residual be 0; therefore, a search for the critical speeds consists of plotting a curve of N vs. R, as shown in Fig. 1.2.

After a trial run if the residual is not 0, two options exist for the next trial run. One option is to increment the speed by the amount previously loaded. This is accomplished by pressing 2nd C′. The other option is to enter a new trial speed. This is accomplished by entering the speed and pressing R/S. The time taken for the calculator to traverse one transfer matrix is approximately 35 s. Therefore, if all six elements are used to describe a model, the time taken to calculate the residual is approximately $3\frac{1}{2}$ min. Since the model does not include damping, the system is in a state of indifferent equilibrium; therefore, all deflections are relative. The program of the third card assumes the slope of the left-hand station to be equal to -1 and then proceeds to calculate the beam deflections moving across the rotor from left to right.

EXAMPLE 1.1

In order to clarify the use of the program a worked example is given.

In reference to Fig. 1.3, the shaft diameter and modulus of elastic-

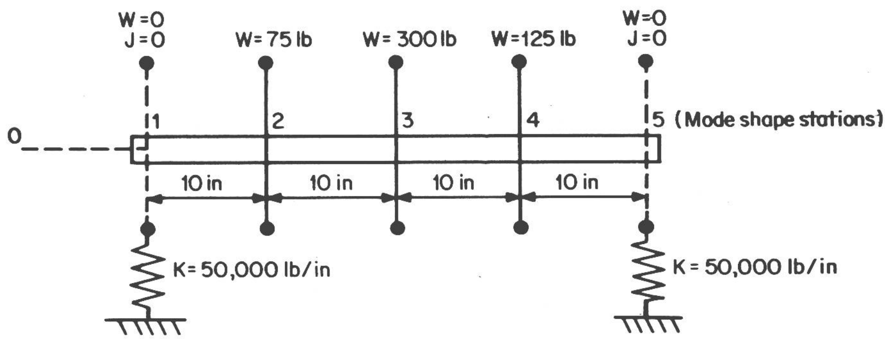

FIGURE 1.3 Rotor diagram for Example 1.1 (*d* is assumed zero in example—i.e., shaft is solid)

ity are assumed to be 1.5 in and 30 × 10^6 lb/in^2, respectively. The disk moment of inertias are assumed to be 0. The problem is to determine the first three modes of vibration of the configuration indicated by Fig. 1.3, together with the shaft mode shapes. The first step in the solution of the problem is to prepare a table of input data.

In Table 1.2 element 1 represents the left-hand bearing; therefore, the element shaft diameter, shaft length, disk weight, and disk moment of inertia are 0. The element spring rate is 50,000 lb/in, which represents the bearing spring rate. If the shaft diameter is set equal to 0, however, the calculator will overflow since in the program the term l/EI is calculated and a zero shaft diameter results in a zero moment of inertia. This can be avoided by setting D equal to unity or in fact any number except 0. This is so because if l is equal to 0, then l/EI will be 0 no matter what the value of D. In elements 2, 3, and 4 the shaft inside diameter is 0 (shaft is solid), and the disk moment of inertia and element spring rate are 0. In element 5 the shaft inside diameter, disk mass, and disk moment of inertia are 0. The data is loaded into the calculator in accordance with the input-output in-

TABLE 1.2 Input Data for Example 1.1

Shaft element number	Shaft outside diameter	Shaft inside diameter	Shaft length	Disk weight	Disk *M* of *I*	Element spring rate
1	1	0	0	0	0	50,000
2	1.5	0	10	75	0	0
3	1.5	0	10	300	0	0
4	1.5	0	10	125	0	0
5	1.5	0	10	0	0	50,000

TABLE 1.3 Critical Speed Search for Example 1.1

N	Residual × 10^{-10}	*N*	Residual × 10^{-10}	*N*	Residual × 10^{-10}
200	+364.14	3500	−320.44	6550	997.239
500	+180.46	3550	−122.65	6580	225.065
680	+2.44	3579	−3.246	6588	12.6315
682	+0.210	3579.8	+0.096	6588.5	−0.7377
685	−3.149	4000	+2095.48	6589	−14.188
1200	−718.89	5000	+8503.31	6590	−40.91
2000	−1878.18	6000	+9430.44	6600	−311.24
3000	−1730.95	6500	+2200.65	7000	−15,174.78

structions. With the data loaded in storage and the program steps of card 2 entered, a search for the critical speeds can begin. This may be accomplished by starting with an assumed speed and then incrementing this speed by a set value until the residual changes sign. Alternatively, the speed may be changed to a value based upon the

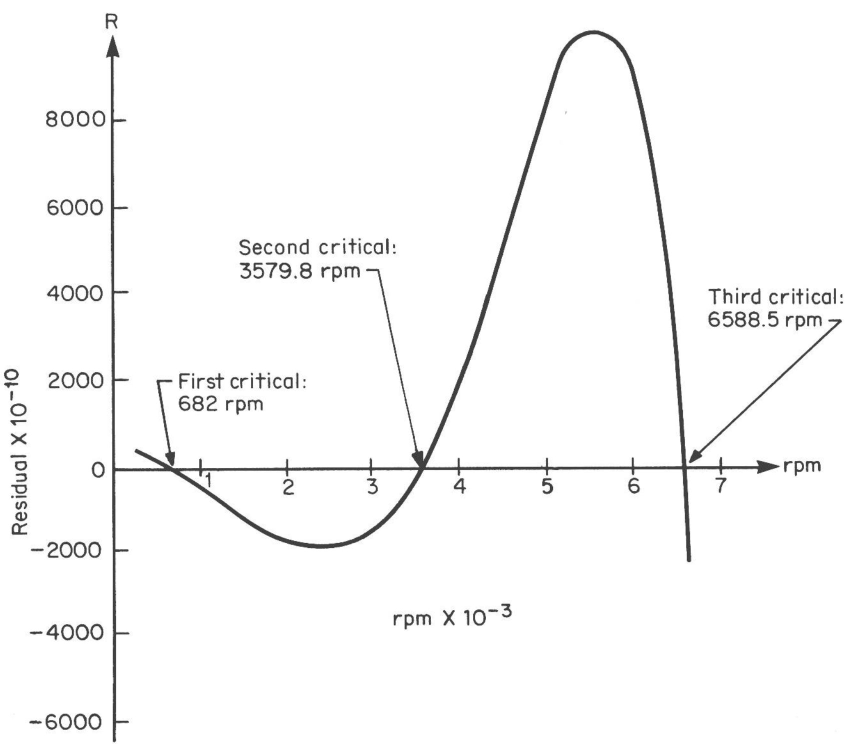

FIGURE 1.4 Plot of residual vs. rotor speed for Example 1.1. (Plotted for illustration purposes only; it is unnecessary to plot the curve in an actual critical speed search.)